Technical Memorandum No. 33-87

Test Report on Sleeve Bearings Made of DU Material

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PREFACE

The studies reported upon herein were performed from August through December of 1961 under National Aeronautics and Space Administration Contract No. NASw-6. The studies are published under National Aeronautics and Space Administration Contract No. NAS 7-100.

ABSTRACT

DU bearings were subjected to series of tests in which only the load was varied while the vacuum, shaft size and finish, and speed were held constant. It was found that load capability is markedly greater in normal atmosphere than in vacuum. It is thought that the mechanism of failure of heavily loaded DU bearings in vacuum is a combination of extreme local heating and bearing-bronze galling on the shaft. For a hardened ⅜ in. diam. stainless steel shaft with 2.5–4.5 rms μ-in. finish rotating continuously at 100 rpm in vacuum, the maximum desirable load is 700 psi, although 250–350 psi may be desirable for design objective.

I. INTRODUCTION

This report covers an investigation of sleeve bearings made of "DU," a proprietary material used in mechanisms for the *Mariner R* radiometer and high-gain antenna yoke. DU (distributed by the Garlock Corporation) was chosen because it appeared to have a higher possibility of success for this particular job than did a number of other promising commercial dry-lubrication sleeve bearings. Dry sleeve bearings are attractive for spacecraft applications because they do not require oil and because it is believed that sleeve bearings can be manufactured with more consistent reliability for each individual bearing than can ball bearings. Sleeve bearings are also more resistant to vibration damage than are ball bearings. On the other hand, sleeve bearings are usually at a disadvantage where initial clearance and wear are important to hold critical alignments.

II. DISCUSSION

A. The DU Material

The manufacturer states that the material known as DU consists of a tin-plated steel backing on which is sintered a thin lining of bronze spheres surrounded by a mixture of TFE fluorocarbon plastic and lead. There is a thin surface layer of the same TFE-lead mixture over the top of the bronze spheres. The TFE-lead impregnated bronze comprises two completely interlocked sponge-like networks: one formed of bronze, the other of the TFE-lead mixture (see Fig. 1).

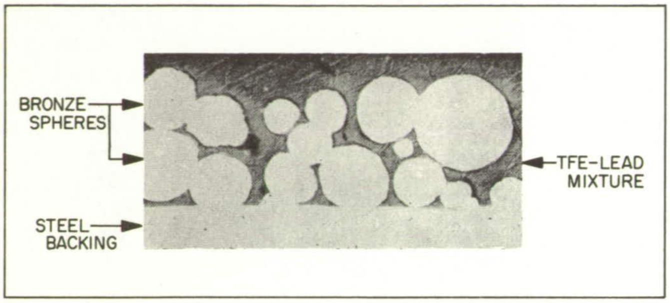

Fig. 1. Cross section of DU bearing material

B. How DU Works

According to the manufacturer, DU is a material composited to realize the advantages of each individual component material, without all their separate disadvantages. While Teflon has extremely favorable low-friction qualities, it also has unfavorable cold flow properties together with poor heat conduction. The lead powder (approximately 20%) in the TFE fluorocarbon and the bronze spheres both help to conduct heat to the steel backing plate. The bronze also serves to bond the Teflon mechanically in place and to inhibit its cold flow.

The manufacturer explains that during operation (in atmosphere), small-area contacts occur between the shaft and the peaks in the porous bronze. The heat developed makes the TFE-lead mixture expand, causing it to smear over the contact area, thus providing lubrication.

C. The Bearing Tester

A low-speed bearing tester was developed to operate—motor included—in a vacuum chamber. The tester consisted of a Superior Electric Company Slo-Syn motor (110 v, 60 cycle, 72 rpm) driving a Gilmore Timing Belt which, in turn, drove the test shaft. The test bearing was loaded radially by a coil spring. The reaction load was carried by two straddling line bearings which were also made of DU material. The motor was converted for vacuum use by replacing the conventional oil lubricated ball bearings with DU bearings. Thus, included in the motor, test shaft, and test bearing, there were bearing loads of three different orders of magnitude: pounds per projected square inch, 100's of psi, and 1000's of psi. This spread of loads shortened the exploratory testing phase. See Appendix B for a diagram of the basic setup, and Fig. 2 for a photograph. Also, see JPL Drawing J8100006 for tester details and sketch B64399 for motor vacuum-conversion instructions.

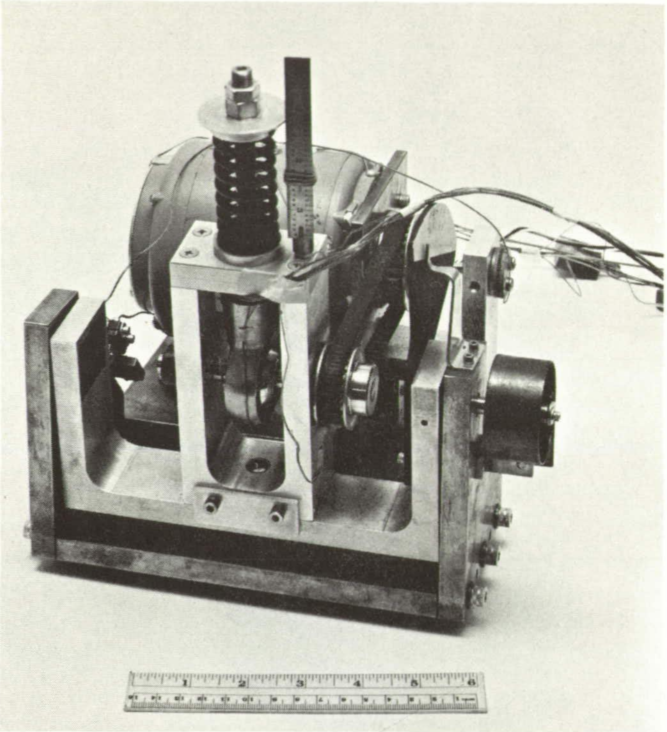

Fig. 2. The bearing tester

D. Thermal Gradients

Because galling between the bronze in the bearing and the steel shaft could be aggravated by high temperature, thermocouple installation was as complete as feasible. Shaft-temperature sensing elements for both the motor and the test shaft were provided. Thermocouples were also attached to detect heat transmission across: (a) the light fit of the test bearing with its insert; (b) the push fit of the insert in its housing; and (c) the running fit of the self-aligning ball joint. Figure 3 is a photograph of the test bearing thermocouples.

As seen in Appendix A, the temperature difference across the inserts was not sufficient to cause design constraint. Temperature proved to be no problem in this test.

E. Test Explanation

In the last (826 hr) run, the test bearing apparently was loaded enough to cause the sintered bronze in the DU bearing to gall slightly on the shaft. Squeaking sounds (transmitted via the chamber base plate) were frequently heard. They were probably caused by bronze to steel galling. This explanation accounts for the increase in torque that always accompanied the squeaks.

The events might have occurred as follows: As each bronze promontory contacted the shaft, it heated up the adjacent Teflon. This permitted the Teflon to flow over the bronze, reducing galling, with concomitant lowering of torque. When the bronze peak wore down enough, the squeaking would completely cease, until another bronze promontory wore through its Teflon cover. Then the same sounds with the simultaneous increase in torque would occur.

Wear on the test bearing was seven times greater than that on the outside bearings, yet the unit load was only four times greater.

In view of the aforementioned sounds, changes in torque, and the rapid wear rate, it appears that the maximum unit loading for a DU bearing used in vacuum at continuous speeds in the neighborhood of 10 surface feet per minute is 700 psi.

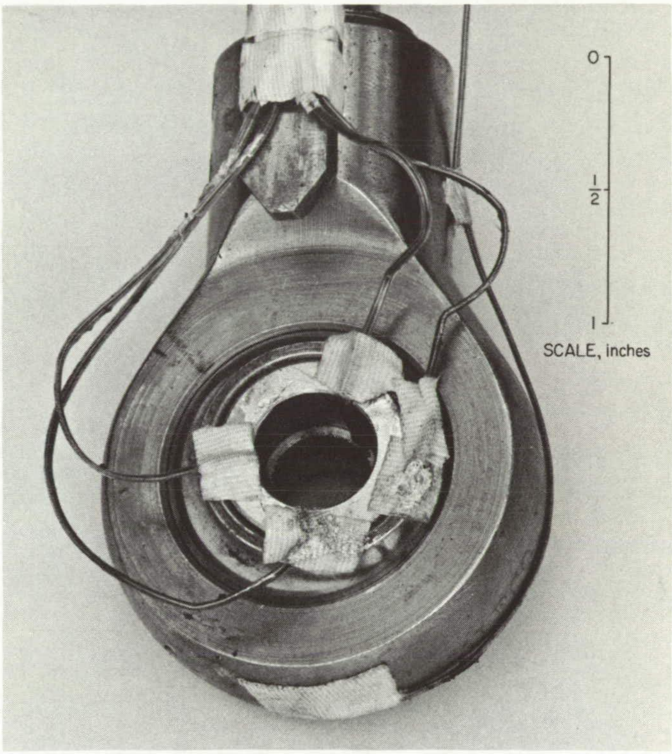

Fig. 3. The bearing installation with thermocouples

III. CONCLUSIONS AND RECOMMENDATIONS

A. Conclusions for DU

1. For continuous operation at 10 ft/~~sec~~ min using a hardened, smoothly ground, 3/8-in. diam., stainless steel shaft, the threshold unit load for vacuum use is 700 psi. Threshold unit load is defined as that load above which the apparent galling takes place at frequent intervals. The allowable frequency must be determined for each application.

2. The highest desirable load for continuous operation appears to be 350 psi, with 250 psi a safer, yet reasonable, number. It should be noted that this is only 10% of the design load in normal atmosphere.

3. Alignment of bearings was found to be important. The line bearings, which are two in. apart, could not tolerate a colinear misalignment of 0.0013 in.

B. Recommendations for DU

1. Conduct further tests with all parameters held constant except the shaft finish. Of particular interest is the influence of the shaft finish on wear rate in vacuum. The range of finishes from 5 to 16 rms μ in. should be checked. With a rougher shaft, results may be better because the lead (a lubricant) may be held better on a rough surface than a smooth one.

2. Conduct further tests to evaluate the effect of speed on life in the very low speed ranges. Of primary interest is the range from 0.1 to 10 rpm.

3. Conduct tests to determine the effect of very small (under 0.0005 in.) diametral clearance on wear.

4. Conduct a sufficient number of tests to determine statistically the life expectancy of DU bearings in vacuum.

C. Conclusions for the Bearing Tester

1. The bearing-tester motor, converted from a standard commercial item to vacuum use by JPL, was proved to be capable of driving miscellaneous test gear in vacuum with a high degree of reliability.

2. The bearing tester's timing belt (made of Hypalon plastic) was found to be useful in the transmission of mechanical power in vacuum.

D. Recommendations for the Bearing Tester

1. The bearing tester could be improved by providing for more accurate torque reading.

2. The tester could be improved by installing a dial indicator in a manner to permit direct reading of test-bearing wear without stopping the test to disassemble the unit.

APPENDIX A

I. Test Objectives

1. Phase I

Exploratory tests on the bearing tester.

a. Learn the techniques of DU bearing installation.

b. Learn how to operate the bearing tester.

c. Proof-test the tester and modify it to ensure reliable and efficient operation.

2. Phase II

Exploratory tests on the DU bearings.

a. Find out under what circumstances DU fails.

b. In the first tests, find where "zero" is on the DU load-life curve in vacuum. To start, apply sufficiently high psi to assure failure within a working day.

c. With each succeeding test, reduce the load until it is reasonable to predict a load which will yield a 1000-hr bearing life.

d. Make initial observations of torque, temperature gradients, vibration, and wear as a function of time.

3. Phase III

Test run.

a. Run a bearing for 1000 hrs at the highest load predicted possible.

b. Gather thermal gradient data at the beginning of the test.

c. Monitor torque, vibrations, sounds, and amperes throughout the test.

II. Test Operation

1. Test 1

This test explored the bearing tester to learn how to operate, trial test, and de-bug it. The most important observation was of a 0.0013-in. misalignment of the line bearings. This condition was corrected.

2. Test 2

This test validated the tester in a seven-hr. combined atmosphere and vacuum test. Also conducted was a study of DU bearing-material wear characteristics in vacuum, when loaded as specified by the manufacturer for 1000 hours of operation in atmosphere. It failed in vacuum in less than one working day. By definition, failure occurred when the torque increased sufficiently to cause the yoke (bearing holder) to hit a mechanical stop. The yoke contains the test bearing, and is itself pivoted on bearings, yet is restrained from motion by a clock spring. See Fig. 2 for details of the tester.

3. Test 3

This test was divided into a number of runs in which all parameters were held constant except load. Load was reduced with each succeeding run until the threshold load was found.

a. Run 1; 2140 psi. This load is claimed by the manufacturer's publications to be normal for at least 1000 hours of operation in atmosphere. It was run at atmospheric pressure until the temperature plateau (138°F) was reached; this took less than two hours.

b. Run 2; 2140 psi. Same load as Run 1, but at pressures between 10^{-4} and $7 \cdot 10^{-6}$ mm Hg. The test was stopped because of excess vibration and yoke oscillation within a half working day.

c. Run 3; 1775 psi. Under vacuum, vibration started immediately, but in atmosphere motion was smooth. It was observed that torque increased about 8% when moved from atmosphere to vacuum. It reduced to the original value when returned to atmosphere.

The motor, when disassembled, was found to be clogged with metal particles. Since the particles were most likely caused by the rotor scraping against the stator, the rotor diameter was machined down 0.009 in. The motor subsequently ran without trouble.

d. Run 4; 1060 psi. Under vacuum, vibration and yoke oscillation commenced immediately. It now appeared that the load was still far in excess of what would be found desirable for 1000 hours. Rather than reduce the load in increments of 25 pounds as originally chosen, another faster way became attractive. During all the preceding tests, when the yoke was oscillating from various maximums to a minimum, the minimum torque value

appeared to be about the same in each instance. Therefore, it seemed a logical assumption to run it at the minimum torque it naturally sought.

Because the coefficient of friction appeared to be about 8% higher in vacuum than in atmosphere, the load was adjusted in atmosphere to give a torque that was correspondingly less than that wished in vacuum.

The bearing ran seven hours prior to failure. The shaft was then removed from the tester, inspected, and color photographed (see Fig. A-1). The dark rings were deposited by the bearings. The smaller deposit in the middle is from the test bearing while the larger, outside pair, are from the line bearings that straddle the test bearing. The middle deposit has a red tinge from copper galled out of the test bearing. By observation, one deducts that the load on the line bearings would be satisfactory for long life while the load on the test bearing is too high. Somewhere between the two loads—270 to 1060 psi—is the threshold load. A load of 700 psi was chosen to try for the 1000-hr. test run.

e. Run 5; 712 psi. This test ran for 826 hours when a building power failure caused the test to terminate. Throughout the test the system coefficient of friction varied between 0.08 and 0.16. When at the low friction-coefficient, the tester ran quietly, but when at the high coefficient, squeaking sounds were audible ten or even twenty feet away from the vacuum equipment. Intermediate values of coefficient of friction were accompanied by squeaks with corresponding degrees of audibility. The silent-squeak-silent cycle occurred on about three distinct occasions, but the same sound less distinct (at times a stethoscope was necessary to detect it) occurred about ten times, with "silence" prevailing only 25% of the time. See Appendix B for the Test 3, Run 5 data sheet.

There was also a deep rumbling background sound that was clearly distinct from the sharp high squeaks. While an explanation is given for the squeaks in the test explanation section, none is given for the deeper sounds.

See Fig. A-2 for a photograph of the shaft after Test 3, Run 5.

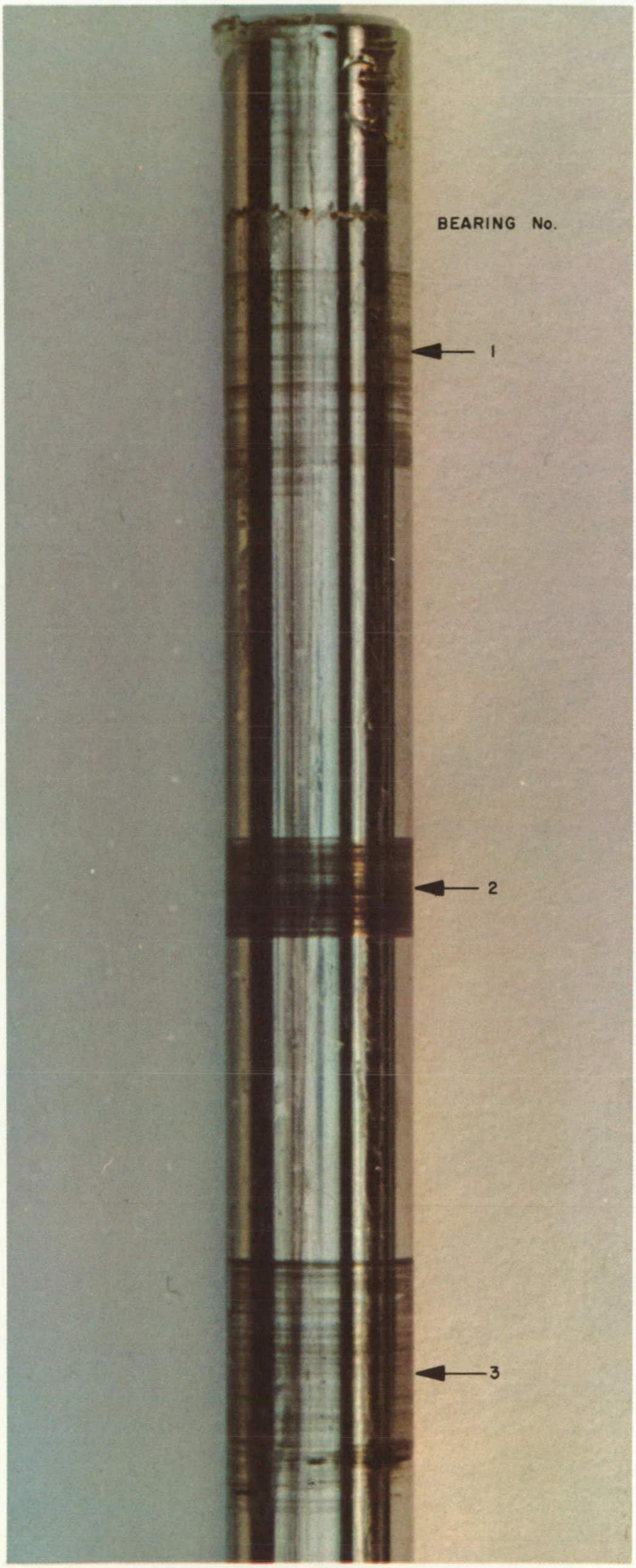

Fig. A-1. Bearing shaft after Test 3, Run 4

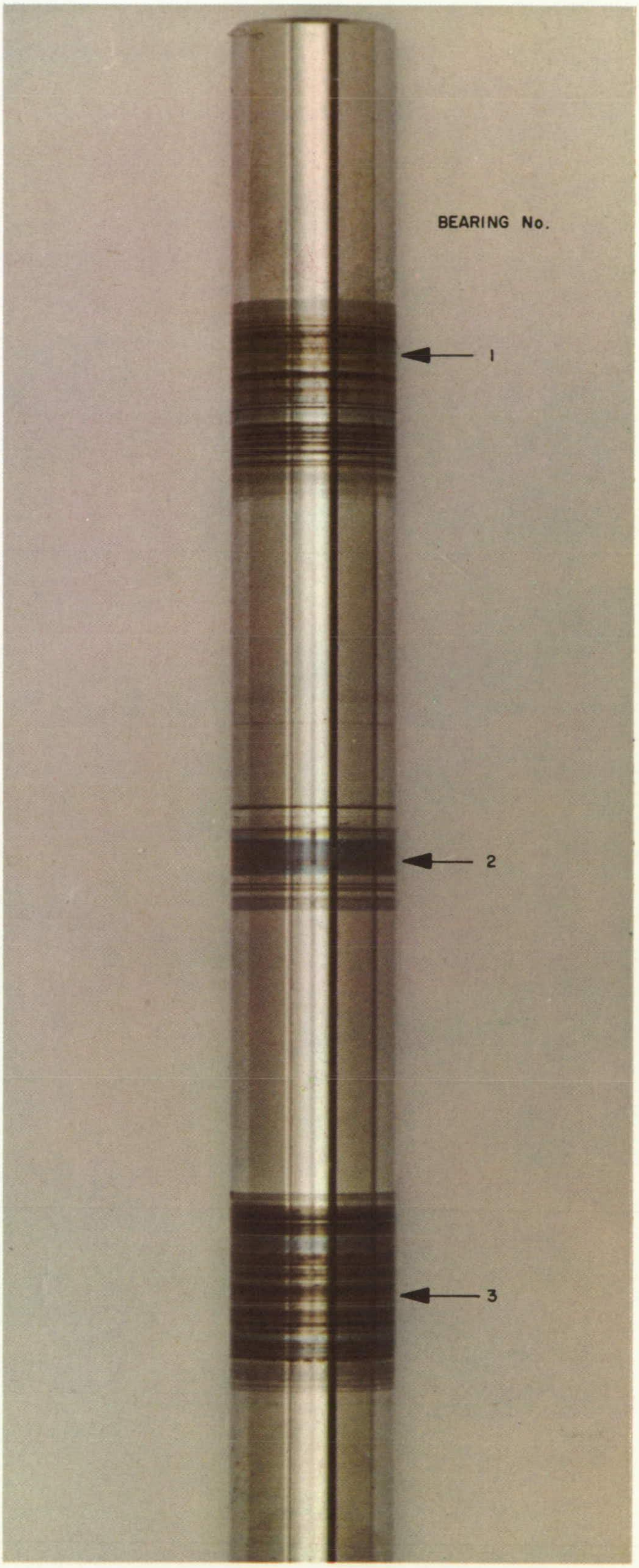

Fig. A-2. Bearing shaft after Test 3, Run 5

APPENDIX B

Data Sheet for Test 3, Run 5

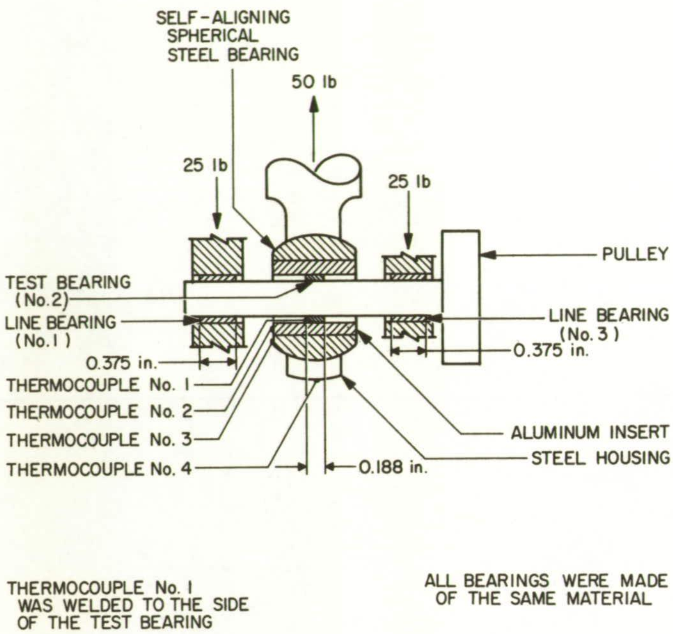

Fig. B-1. Bearing-tester setup

Test started	2 Nov. 1961
Test terminated	7 Dec. 1961
Total running time	826 hrs
Pressure	5×10^{-6} mm Hg
Ambient temperature	70°F (room)
Shaft	
Size	⅜-in. diam nominal
Material	AISI 440C stainless
Heat treat	277,000 psi
Finish	2.5 to 4.5 rms μ in.
Speed	
rpm	100
Surface	9.9 ft/min
Radial force	
Total load	Line bearing: 25 lbs Test bearing: 50 lbs
Unit load	Line bearing: 178 psi Test bearing: 712 psi
Initial clearance	
Bearing No. 2	0.0007 in. on the diam
Bearing No. 1	0.0010 in. on the diam
Bearing No. 3	0.0009 in. on the diam

Running data

Torque: Varied at random between 1.5 and 3.0 in.-lbs throughout the test. High torque was accompanied by squeaking sounds.

System coeffiient of friction

Varied from 0.08 to 0.16 throughout the test.

Thermal gradient

Thermocouple No. 1 (bearing wall): 133°F
Thermocouple No. 2 (aluminum insert): 133°F
Thermocouple No. 3 (spherical bearing): 132°F
Thermocouple No. 4 (housing): 128°F

Note: Interfaces were clean "as fabricated"; they were dirty only from normal shop handling.

Bearing wear on radius in direction of load

Bearing No. 2: 0.0014 in. on the radius
Bearing No. 1: 0.0001 in. on the radius
Bearing No. 3: 0.0001 in. on the radius

Wear rate on the radius

Bearing No. 2: 1.69×10^{-6} in. wear/hr
Bearing No. 1: 1.2×10^{-7} in. wear/hr
Bearing No. 3: 1.21×10^{-7} in. wear/hr

Wear rate per unit load on the radius

Bearing No. 2: 2.38×10^{-9} in. wear/hr/psi
Bearing No. 1: 6.8×10^{-10} in. wear/hr/psi
Bearing No. 3: 6.8×10^{-10} in. wear/hr/psi

Note: The ratio of the wear rate per unit load of the test bearing (No. 2) to either line bearing (No. 1 or No. 3) is 3.5 to 1. The high wear rate of Bearing No. 2, in addition to the copper it deposited on the shaft, indicates that it was wearing in a manner different from that of the other bearings. Hence the conclusion that a load of 700 psi is too high for normal running in a long-duration vacuum operation.